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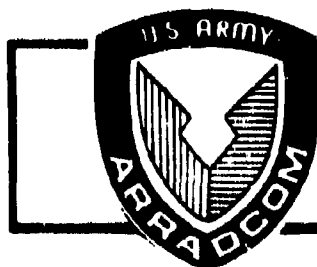
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TECHNICAL REPORT ARLCB-TR-81006

**A CRITICAL REVIEW OF THE DRY FRICTION AND
COMPRESSIBLE FLUID RECOIL/COUNTER-RECOIL CONCEPTS**

H. J. Sneck

February 1981



**US ARMY ARMAMENT RESEARCH AND DEVELOPMENT COMMAND
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20. ABSTRACT (Continue on reverse side if necessary and identify by block number) This report reviews two new recoil systems as possible replacements for the conventional mechanisms currently in use on medium and large size cannon. One uses compressible fluid to dissipate energy in viscous friction and store elastic energy for counter recoil. The other uses coulomb or dry friction for dissipation of the recoil energy.		

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INTRODUCTION

Two novel recoil/counter-recoil systems have been proposed as replacements for the conventional hydrospring or hydropneumatic mechanisms currently in use. The basic operating principles and characteristics of these schemes are described in this report, along with their advantages and disadvantages.

The technologies critical to the success of these new devices are identified. The areas where further development will be required are discussed. Suggestions for future work are made.

RECOIL MECHANISM USING COULOMB FRICTION

Recoil mechanisms must dissipate about 95 percent of the total recoil energy leaving about 5 percent for recuperation and dissipation during counter-recoil. In conventional recoil systems, this dissipation is accomplished by forcing a fluid (usually liquid) through a restriction. Viscous friction generated by the fluid (largely by turbulence) converts the mechanical energy of the recoiling parts into thermal energy within the fluid. This energy is then transferred to the surroundings by conduction, convection and radiation.

Hydropneumatic recoil mechanisms currently in use satisfy the requirements of the basic recoil mechanism described above. Although they are passive systems, they are far from simple, often requiring close fits, accurately sized holes, high pressure seals, special working fluids, etc.

The possibility of dissipating the recoil energy via dry (coulomb) friction rather than viscous has considerable appeal. Such a device, if purely mechanical, could be relatively simple to manufacture, maintain, and replace. Its serviceable life would be a predictable function of the friction material wear as opposed to the statistical estimate of the effective seal

life. It should also be relatively insensitive to variations in ambient temperatures, and have a long serviceable storage life.

General Operating Characteristics

There are no known operational dry friction recoil mechanisms. The closest analogues are automotive and aircraft brakes. Unfortunately the operating conditions of these dissipative devices differ in two significant respects from those which would be imposed in a recoil application.

The characteristic operating time of ordinary braking systems are of the order tens of seconds. Recoil devices must accomplish their dissipating function in tenths of a second, i.e., in two orders of magnitude shorter time. As a consequence exceedingly high normal forces are required to produce the required level of frictional forces. Large forces can be applied by hydraulic systems, but that would add a non-mechanical support system to a device which was conceived as purely mechanical. If the system is constrained to be mechanical, non-hydraulic schemes must be devised to apply these large forces.

The other significant difference is the condition which exists when the friction force is applied. In the case of a braking system, the mating surfaces are moving when contact is made. In the case of a gun the mating surfaces are at rest relative to each other (in battery position) and highly loaded when motion begins. This "start from loaded condition" will impose very severe impulsive shear loads on the friction material, much larger than experienced under ordinary braking conditions. Little or no data appears to be available on the effects of impulsive starts under load conditions. There is no way of knowing, short of testing, how this unconventional abrupt initial loading will affect life of friction materials.

The mechanical performance of a dry friction recoil system has been simulated using a mathematical model which was implemented using a digital computer.⁽¹⁾ Two methods of applying the normal force were considered. In one case a constant normal force was applied yielding a constant friction force. In the other case the normal load was increased linearly with the recoil distance simulating the activation of the normal force by the recuperating spring. The performance of these systems were compared with that of a typical hydropneumatic mechanism.

The recoil performance of the hydropneumatic system was found to be superior to both of the mechanical systems. The reason for this is that in the hydropneumatic system the rate of dissipation is proportional to the cube of the instantaneous recoil velocity, while in the dry system with a constant normal force it is proportional to the first power of the velocity.

Using the recuperator spring to activate the normal force (increasing it linearly with recoil distance) was considered because it was thought that it might provide a means of compensating the dry system for this difference. Unfortunately this scheme does not produce the largest frictional forces at the beginning of recoil where they are most needed. As a result, its performance was inferior to the constant normal force system.

The hydropneumatic recoil is ideally suited to its purpose since it develops the highest resisting forces during those portions of the recoil stroke where they are required. The highly non-linear relationship between velocity and dissipation provides the very sensitive internal feed-back mechanism of this system.

The following table compares the two systems:

TABLE I - System Characteristics

System	Friction Force	Rate of Energy Dissipation
Hydropneumatic	v^2	v^3
Dry friction	N	NV

V = recoil velocity

N = normal force

The only way the two systems can be made comparable is by "programming" the normal force N to vary during the recoil stroke.

There are two readily available signals which could be tapped to control the normal force. These are the recoil displacement and the recoil acceleration. Linking the normal force to the counter-recoil spring was an unsuccessful attempt to feed-back the displacement. Another, and more flexible way of introducing this coupling, would be to actuate the normal force with a cam/follower mechanism. The cam could be attached to either the grounded or recoiling members and the follower with force amplifying linkage attached to the other. With such a mechanism it should be possible to vary N according to any pre-determined displacement related profile. Ideally N should be sufficient to maintain battery position at rest, increase very rapidly during the early stages of recoil, and decrease near the end of the stroke. Since the same cam profile would be followed during counter-recoil it would be necessary to reduce the "in-battery" pre-load at the end of the recoil and restore it at the end of the counter-recoil. A parallel cam/ratchet system could perform that function.

The energy for restoring the required "in battery" pre-load could be taken from the counter-recoil kinetic energy.

Although the displacement feed-back scheme described above may not be particularly simple to execute, it is certainly the most direct. The normal force could also be activated by a mass, attached to the recoiling part, which is allowed to move a small limited distance relative to the recoiling part. This mass would provide a reactive force roughly equal to the product of its mass and the instantaneous acceleration of the recoiling parts. The reactive force could then be used to vary the normal force. Unfortunately when the friction force is entered into the equation for the recoil dynamics it will combine with the recoil acceleration, times the very large mass of the recoiling parts, with the result that the effective mass of the recoiling part will be increased. The following equations illustrate this effect.

The equation of motion is

$$(M + m) \ddot{x} + \mu N + Kx = f_a(t)$$

where

M = original recoiling mass

m = added activator mass

μ = coefficient of friction

N = normal force

K = recuperator spring coefficient

$f_a(t)$ = force generated by the internal ballistics

If $N = B \dot{x}$ where B is the mechanical amplification factor (provided by linkages, cams, etc), then the equation of motion becomes

$$[M + (1 + B\mu) m] \ddot{x} + Kx = F_a(t)$$

The effect of the friction force generated this way could certainly be obtained much more simply by attaching a mass of $(1 \times B\mu) m$ to the recoiling parts and eliminating the friction mechanism altogether.

To create a velocity-dependent normal force would require the displacement signal or the mechanical integration of the acceleration signal, neither of which would seem to be practical. The alternative would be to activate the normal force with a dashpot rather than a spring. The normal force would then be dependent on the velocity squared just as in the hydro-pneumatic system. The resulting recoil system would, however, be a hybrid dry friction/hydropneumatically activated one, rather than purely mechanical as desired.

This rather brief review of some of the ways in which the normal force might be controlled serves to illustrate the need for further work in the area of self-energizing concepts. Some other possibilities are considered in reference (1).

Materials

The literature contains a wealth of information on the wear and frictional characteristics of braking materials. Carefully performed tests⁽¹⁾ show that contamination in the form of dry particulate or liquid (water, grease, hydraulic fluid) can greatly reduce the friction coefficient especially on the first application of a load. Consistent performance of the dry recoil mechanism can be attained only if all contamination of the sliding surfaces is avoided. To achieve this it is obvious that these surfaces must be enclosed and sealed from outside contaminants. This would seem to be a relatively simple matter considering that the sealant need not withstand any significant pressures.

There is, however, an added complication. It is known that about 90 percent of the heat generated by braking systems goes into the mating material. Adequate cooling of these surfaces (e.g. ways) requires either direct exposure to the ambient or an internal auxiliary cooling system (e.g. a coolant pumped through the mating material). For simplicity direct radiative and convection cooling would be preferred. On the other hand, if the system must be encapsulated to prevent contamination, direct cooling may be very difficult to achieve. The need for adequate cooling could very well be the decisive factor in the choice between competing self-energizing designs.

In addition to the high starting impulsive forces and large operating loads already mentioned, the friction materials must also be able to withstand exceedingly high temperatures while at the same time exhibiting very little "fade". They must be both corrosion and wear resistant. They cannot generate toxic gases or residues. All these conditions must be met with readily available, relative inexpensive materials. Failure to find or compound such materials will prevent the realization of any dry-friction recoil concept.

Assessment

Probably the most crucial factor in the success of the dry-friction concept are the materials. The friction material requirements for this application are much more severe than those for even the worst braking applications. For this reason it will be necessary to devise tests for repetitive "under-load" impulsive starts followed by short duration high load sliding. For the tests to be valid the cooling conditions of the tested material must duplicate what is practical in field operation.

The mechanical design of the self-acting normal force applying mechanism would seem to be relatively straight forward and dependent to some extent on the friction material limitations. There is considerable room for invention and innovation in this area, with many alternatives possible. As an example the material problems associated with impulsive starting loads might be alleviated by quickly reducing the in-battery holding force just before firing. There are probably many ways this function can be performed automatically, simply, and reliably.

The constraints on the designs are the usual ones, reliability, simplicity, serviceability and size. Initial designs should be simulated mathematically for their dynamic and thermal responses before any prototypes are constructed. Material testing could proceed in parallel with the concept modeling, the two being mutually supportive.

Since the dry-friction recoil is now little more than a concept its development should be viewed as a long-range project which will require considerable developmental effort and testing before the concept can be proven practical or not. The highest priority should be assigned to the material selection, screening and testing where the uncertainty of success is the highest.

COMPRESSIBLE FLUID RECOIL MECHANISMS

Principle of Operation

The compressible fluid recoil mechanism stores recuperating energy during the recoil stroke by slightly reducing the volume of the oil trapped between the piston and cylinder. This compression is accomplished by drawing a stepped shaft through a relatively rigid cylinder as shown in the sketch.

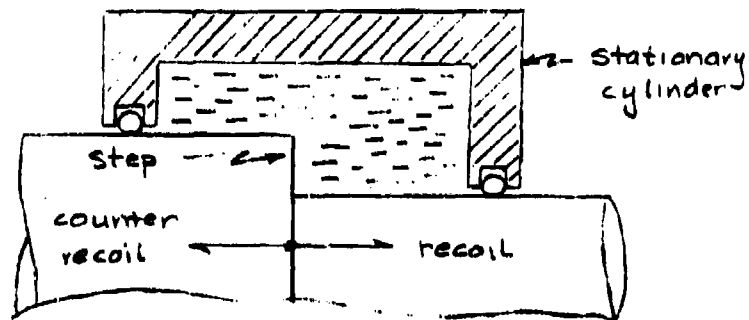


Figure 1

An analysis of this simple system (see Appendix) shows that the pressure rise is approximately proportional to the tube displacement.

The device sketched above is completely reversible. The pressure rise during the recoil stroke is exactly repeated on the counter-recoil stroke because there is no energy dissipating mechanism. A throttling orifice is added to the reversible device to provide the required dissipation. Buffering at the end of counter-recoil is also accomplished by a throttling mechanism. Figure 2 shows these energy dissipating devices added to the reversible system.

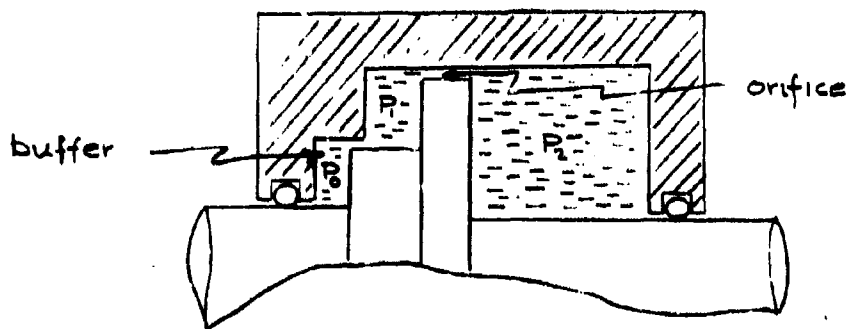


Figure 2

During the recoil stroke of this device pressure P_2 is always larger than P_1 due to the throttling effect of the orifice on the fluid passing from region 2 to 1. Pressure P_2 acts on a larger axial area than P_1 so that there is a net force generated by the orifice which also opposes the recoil stroke. This force is very nearly proportional to the square of the recoil velocity (See Appendix).

As the recoil velocity slows near the end of the recoil stroke pressures P_1 and P_2 tend to equalize. When the end of the stroke is reached the pressures will be very nearly those which would have occurred in the reversible device. This pressure, which results from slightly compressing the fluid, acts on the net area of the step providing the restoring force to initiate counter-recoil. The energy which was not dissipated by the orifice flow during recoil is stored in the compressed fluid which acts like a spring.

Once counter-recoil motion begins fluid will move through the orifice from contracting region 1 to expanding region 2. To accomplish this liquid transfer P_1 must be larger than P_2 . The force accelerating counter-recoil is now less than at the beginning of the counter-recoil stroke. It is convenient to think of this accelerating force as being composed of two components. These can be identified by separating the counter recoil forces (shown in Figure 3).

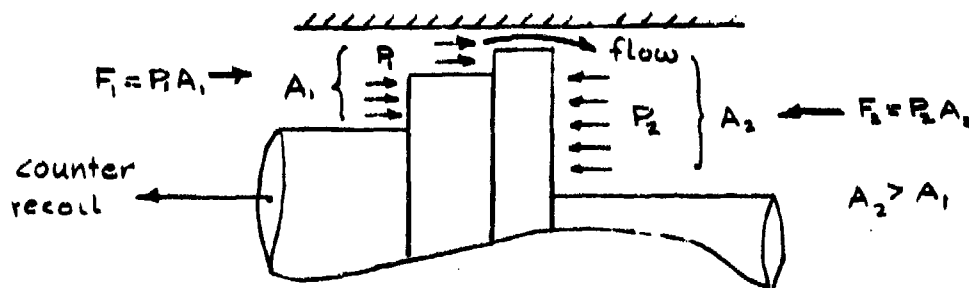


Figure 3

into two parts

$$F_2 - F_1 = P_2 A_2 - P_1 A_1 = P_2 (A_2 - A_1) - (P_1 - P_2) A_1$$

The force $P_2 (A_2 - A_1)$ is the counter-recoil accelerating force due to liquid pressurization, i.e., the elastic energy stored in the liquid (and the cylinder walls). It is dependent on the position of the step and not its velocity. It is comparable to, but somewhat smaller than, the force which would have been generated without an orifice (see Figure 1).

The force $(P_1 - P_2) A_1$ is the counter-recoil decelerating force. The difference $P_1 - P_2$ is determined by the liquid flow-rate through the orifice from region 1 to region 2. It is dependent on the orifice geometry and is again roughly proportional to the counter-recoil velocity squared.

During the recoil stroke both of these forces oppose the motion tending to decelerate recoil. During counter-recoil the elastic compressed liquid force $P_2 (A_2 - A_1)$ acts to accelerate the stroke, while $(P_1 - P_2) A_1$ acts to decelerate it.

The dynamics of this system are modeled by the equation

$$m\ddot{x} + b(x)|\dot{x}| + (a + kx) = f_a(t) + W_x$$

where

m = mass of the moving components

b = damping coefficient, dependent on the orifice geometry and the fluid properties (see Appendix)

k = spring coefficient, dependent on the geometry of the design and the fluid/cylinder elasticity (see Appendix)

$f_a(t)$ = forces applied to the system by the internal ballistics of the gun

a = pre-load coefficient

W_x = component of the gravity load

In this equation $x = 0$ corresponds to battery position. The elastic force $(a + Kx)$ is written with a pre-load coefficient (a) to emphasize the need for a residual fluid pressure in the battery position ($x = 0$) to support gravity loads. The specification that $a > 0$ corresponds to saying that $P_2 > 0$ when the tube is in battery position.

For a given gun (with m , $f_a(t)$, and W_x) the system response x is governed by the coefficients a , b , and K . These coefficients are determined by the geometry of the system, the fluid properties, and the pre-load pressure. The solution of this equation or the more complicated versions which would result from causing b and K to vary with x (by using tapered bores and buffers) cannot be solved by classical methods. Its solution using a computer is, however, relatively straightforward.⁽²⁾

Comments on the Concept

The concept is fundamentally sound and simple. Its practical implementation raises a number of questions which require careful consideration. These will be discussed in the following paragraphs:

1. Return to Battery

The unique feature of the system is elimination of large moving elastic mechanical energy storing elements such as springs. However, the system must be pre-loaded (just as with springs) to maintain the tube in battery position when not firing. Early designs attempted to avoid a pre-load force by using a latch (Chrysler version) to hold the tube in battery position.⁽³⁾

This scheme proved unreliable and many rounds failed to return to battery. It was found that the problem could be overcome if the pressure relief valve on the cylinder was not permitted to release the fluid pressure near the end of the counter-recoil stroke. The residual pressure retained apparently provided the required pre-load.

Present designs incorporate a means for intentional hydraulic pre-loading. These systems, unfortunately, require additional devices (accumulator, spring, hydraulic pump with check valve, etc.) which reduce the simplicity and reliability of the original concept.

2. Purging and Replenishment

Since at least two fairly large high pressure seals are needed, some leakage of fluid from the system is inevitable. Some means of making up these losses is necessary.

The pre-load system mentioned above could also fulfill this function.

It has also been observed that entrained gases accumulate within the system.⁽¹⁾ The source of these gases is not clearly understood. Measurements have shown that sub-ambient pressures are generated in the buffer during the early part of the recoil stroke. It has been suggested that gases are drawn out of solution or drawn in through the seals by these low pressures.

Whatever the source, some provision must be made for purging gases from the system.

Pre-loading the system before and during firing could have the added benefit of suppressing gas dissolution and ingestion through the seals.

3. Seals

Up to this point only co-axial designs have been proposed, apparently based on accuracy considerations. These configurations require high pressure sealing systems at both ends of the cylinder. Because of the relatively large diameter of the sealed surface the peripheral length of these seals tends to be very large. The reliability and ease with which these seals could be replaced is questionable, especially under field conditions. Obtaining easy-to-install and efficient seals could require a major development effort.

4. Material Properties

Consistent performance of the system depends on the constancy of the fluid bulk modulus β and to a lesser extent its viscosity. Both of these properties can vary considerably with temperature.⁽⁴⁾ The choice of the working fluid is therefore strongly dependent on its thermally dependent properties and its compatibility with the sealing materials. Preliminary work has been done to find fluids with relatively large bulk moduli.⁽⁴⁾ Additional work will undoubtedly be required to reduce temperature sensitivity.

Non-Concentric Designs

All the compressible fluid recoil designs proposed so far are concentric configurations requiring two sets of high-pressure to ambient external seals. There seem to be no proposed non-concentric designs which could use one less set of high pressure seals. These deserve some consideration, at least conceptually.

Assessment

The compressible fluid recoil/counter-recoil system has already been carried well beyond the conceptual stage by Chrysler⁽³⁾. Present plans call for testing of a full size advanced, concentric design by the U. S. Army in mid-1981. The results of these tests should provide answers to some of the questions raised above. They may also reveal problems which are, as yet, unidentified.

It is recommended that this program be monitored rather closely since it is anticipated that it will pin-point those areas which require further development. Further consideration of the concept should await the outcome of this test program when a better informed assessment can be made.

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APPENDIX

Analysis of Simple Model

The volume of fluid
at any instant is

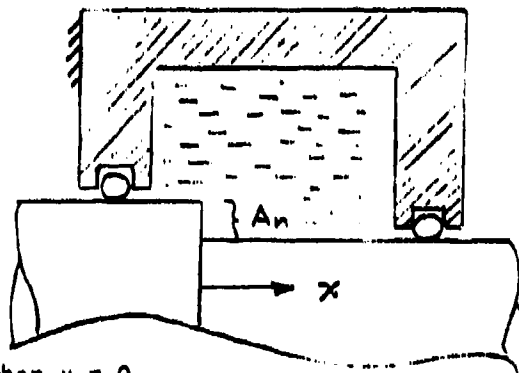
$$V = V_0 - x A_n$$

where

V_0 = original volume when $x = 0$

A_n = step area = $A_2 - A_1$

$x = 0$ in battery position



The mass of the fluid confined within the device is

$$m = \rho V$$

This mass remains constant, so that

$$\frac{dm}{dt} = V \frac{d\rho}{dt} + \rho \frac{dV}{dt} = 0$$

Since ρ varies only slightly, in this equation $\rho \approx \rho_0$

The bulk modulus is defined by

$$d\rho = \frac{d\rho}{\beta} \rho$$

or

$$\frac{d\rho}{dt} = \frac{\rho_0}{\beta} \frac{d\rho}{dt}$$

The first equation yields

$$dV = - \dot{x} A_n$$

Substituting these into the mass conservation equation reduces it to

$$\frac{dp}{dt} = \frac{\dot{x}}{V} (\beta A_n)$$

or

$$dp = \beta A_n \left(\frac{dx}{V_0 - x A_n} \right)$$

Integrating

$$P \Big|_{P_0}^P = -\beta \ln(V_0 - x A_n) \Big|_{x_0=0}^x$$

$$P - P_0 = \beta \ln \left(\frac{V_0}{V_0 - x A_n} \right)$$

Rearranging

$$x = \frac{V_0}{A_n} \left(1 - e^{-\frac{P - P_0}{\beta}} \right)$$

Typical values are

$$\beta \sim 150,000 \text{ psi}$$

$$P - P_0 \sim 3,000 \text{ psi}$$

$$\frac{P - P_0}{\beta} \sim 2 (10^{-2}) \text{ or less}$$

Since $\frac{P - P_0}{\beta}$ is small

$$x \approx \frac{V_0}{A_n} \left(\frac{P - P_0}{\beta} \right)$$

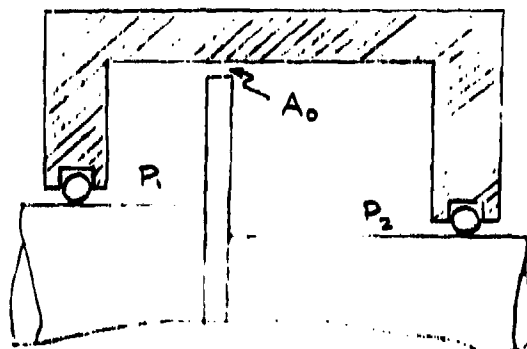
$$\text{or } P - P_0 = \left(\frac{\beta A_n}{V_0} \right) x$$

The opposing force is

$$F = P A_n = P_0 A_n + \left(\frac{\beta A_n^2}{V_0} \right) x$$

$$= a + x$$

Orifice Flow



The flow rate across the orifice is

$$Q = A_n \dot{x} = C A_o \sqrt{\frac{2g_o}{\rho} (P_1 - P_2)}$$

or

$$P_1 - P_2 = \frac{\rho_o}{2g_o} \left(\frac{A_n}{C A_o} \right)^2 (\dot{x})^2$$

The damping force is

$$(P_1 - P_2) A_1 = \frac{\rho_o A_1}{2g_o} \left(\frac{A_n}{C A_o} \right)^2 |\dot{x}| \dot{x}$$

In the turbulent regime expected with this device, C is relatively insensitive to changes in viscosity and density.

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